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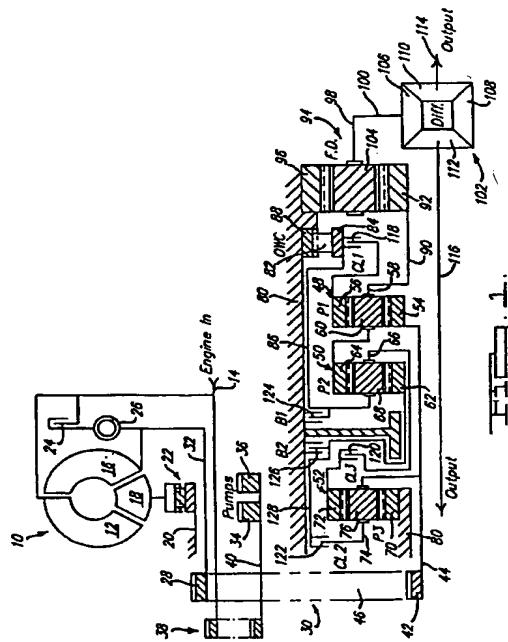
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(54) Multiple-speed automatic transmission for a motor vehicle

(57) A multiple-speed automatic transmission is described for automotive vehicles able to produce three underdrive speed ratios, a near direct ratio, and two overdrive ratios. A kinematic arrangement includes three simple planetary gear units (48, 50, 52). One of these produces an underdrive output drivably connected to components of the other gear units, two friction clutches (118, 122), three friction brakes (124, 126, 140), an overrunning coupling (82), a final drive gearset (94), a differential mechanism (102), a chain drive mechanism (30) for transmitting power from the turbine (16) of a torque converter (10) to the input shaft, which drives elements of the planetary gear units.



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Description**Field of the invention**

This invention relates to the field of automatic transmissions for motor vehicles

Description of the prior art

Transaxle assemblies for use with an internal combustion vehicle engine are known that have a hydrokinetic torque converter and multiple ratio gearing. The invention finds application in a vehicle in which the engine and the transaxle are mounted in a common powertrain package with the axis of the crankshaft of the engine in alignment with the axis of the hydrokinetic torque converter. The multiple ratio planetary gearing is mounted on the output shaft axis, which is arranged in parallel with the converter axis. The engine and output shaft axis may be mounted transversely in the forward vehicle engine and transaxle compartment of a front wheel drive vehicle with forward traction wheels.

Various arrangements of gearing, clutches, brakes, and couplings for multiple-speed ratio power transmissions include two planetary gear units having some of their components mutually interconnected and another planetary gear unit, or pair of meshing pinions and gears, located between the engine and the two planetary gear units. Input speed to the two planetary gear units is provided either through a direct drive connection to the engine or through a gear mechanism that underdrives the input. Therefore, torque applied to the components of the two planetary gear units is greater than it would be if the input were driven at a higher speed. Consequently, those components and the friction elements that control them are larger and heavier than if torque carried by them were lower.

Examples of multiple-speed planetary gear units having a two-speed input are described in U.S. Patents 5,039,305; 5,250,011; 5,267,913; and 5,267,916.

Hydraulically actuated friction clutches require rotary hydraulic seals to minimise leakage and related losses. Since both of the members are drivably connected by a clutch rotate, use of rotary seals is extensive to avoid leakage and the resulting variation of gear shift quality. Furthermore, clutches frequently require balance dams to minimise adverse effects of centrifugal pressure head induced by rotation.

A friction brake connects one rotating member to a nonrotating member, usually the transmission housing. The brake piston does not rotate. Therefore, because a brake requires less hydraulic sealing, it is preferable to minimise the number of clutches required to control operation of the gear mechanism and to substitute, wherever possible, a brake for a clutch.

Summary of the invention

According to the present invention, there is provided an automatic transmission as hereinafter set forth in **Claim 1 of the appended claims.**

The invention enables an automatic transmission to be produced that is capable of producing six forward speed ratios and a reverse drive ratio. The six forward speed ratios include three underdrive ratios, a near direct drive ratio, and two overdrive ratios. The transmission avoids the disadvantages present in the prior art by overdriving input members of the multiple speed ratio planetary gear units in relation to the speed of the input shaft driven by the engine. Furthermore, the number of friction clutches is minimised and the number of friction brakes optimised.

Brief description of the drawings

20 The invention will now be described further, by way of example, with reference to the accompanying drawings, in which:

25 Figure 1 is a schematic diagram of the kinematic arrangement of the gears, clutches, brakes, and couplings for a preferred embodiment of this invention,

30 Figure 2 is a chart that shows the pattern of engagement and release of the clutches, brakes, and couplings required to produce the various forward drive ratios and reverse ratio for the transmission mechanism of Figure 1,

35 Figure 3 is a schematic diagram of the kinematic arrangement of an alternative embodiment of the automatic transmission of this invention, and

40 Figure 4 is a chart that shows the pattern of engagement and release of the clutches, brakes, and couplings required to produce forward drive and reverse drive of the transmission mechanism of Figure 3.

Description of the preferred embodiment

Referring first to Figure 1, a hydrokinetic torque converter 10 includes an impeller 12 connected to the crankshaft 14 of an internal combustion engine. A bladed turbine 16, a bladed stator 18, and the impeller 12 define a toroidal fluid flow circuit, whereby the impeller is hydrokinetically connected to the turbine. The stator 18 is supported rotatably on a stationary stator sleeve shaft 20, and an overrunning brake 22 anchors the stator to shaft 20 to prevent rotation of the stator in the direction opposite the direction of rotation of the impeller, although freewheeling motion in the opposite direction is permitted.

The torque converter assembly includes a lock-up clutch 24 located within the torque converter impeller housing. Arranged in series with the lock-up clutch 24

between crankshaft 14 and turbine wheel 16 is a damper 26, which is drivably connected both to the turbine wheel and to a sheave or sprocket wheel 20 of a chain drive mechanism 30 through a sleeve shaft 32 arranged parallel to crankshaft 14. The damper absorbs transient torque fluctuations associated with engagement of the lock-up clutch 14. Fluid to the torque converter is supplied from the output of hydraulic pumps 34, 36, which are driven through a drive mechanism such as a chain drive 38 having a sheave or sprocket wheel drivably connected to crankshaft 14 and another sheave or sprocket wheel connected to the input shaft 40 of the pumps.

The chain drive mechanism includes a sheave 42 fixed to a transmission input shaft 44, which is drivably connected to elements of several planetary gear units arranged coaxially with shaft 44. Power is transmitted between crankshaft 14 and input shaft 24 by a drive belt or chain 46, which engages the sprockets or sheaves 28, 42.

Planetary gearing includes first, second, and third planetary gear units 48, 50, 52. The first gear unit 48 includes a sun gear 54, ring gear 56, pinion carrier 58, and a set of planet pins 60 supported rotatably on carrier 58 in meshing engagement with sun gear 54 and ring gear 56. The second gear unit 50 includes a sun gear 62, ring gear 64, pinion carrier 66, and a set of planet pinions 68 rotatably supported on carrier 66 in meshing engagement with sun gear 62 and ring gear 64. The third gear unit 52 includes a sun gear 70, ring gear 72, pinion carrier 74, and a set of planet pinions 76 rotatably supported on carrier 74 in meshing engagement with ring gear 72 and sun gear 70. Carrier 74 and sun gear 54 are drivably connected directly to input shaft 54. Ring gear 70 is permanently fixed against rotation on the transmission housing 80.

One-way brake 82 includes an inner race 84 drivably connected by member 86 to carrier 66, an outer race 88 fixed against rotation on the transmission housing 80, and a set of drive elements located in the annular space between the inner race and outer race adapted to complete a one-way drive connection between the races and to allow the inner race to turn with respect to the outer race in the opposite direction of rotation.

Carrier 58 is continually drivably connected to ring gear 64 and is connected through output sleeve shaft 90 to the ring gear 92 of a final drive gearset 94, which includes a ring gear 96 permanently fixed against rotation on the transmission housing, pinion carrier 98 connected to spindle 100 of an axle differential mechanism 102, and a set of planet pinions 104 rotatably supported on carrier 98 in meshing engagement with ring gear 96 and sun gear 92. Spindle 100 rotatably drives bevel pinions 106, 108, which are in continual mutual driving engagement with side gears 110, 112, respectively drivably connected to right-hand and left-hand drive shafts 114, 116.

Forward clutch (CL1) 118 is a fluid-actuated, friction

clutch whose engagement drivably connects carrier 66 and ring gear 56. Clutch (CL2) 122 drivably connects and releases carrier 74 and sun gear 62. Clutch (CL3) 120 drivably connects and releases carrier 66 and ring gear 72. Brake (B1) 124 establishes and disestablishes a drive connection between member 86 and the transmission casing, thereby holding against rotation and releasing carrier 66, ring gear 56, and race 84. Brake (B2) 126 establishes and disestablishes a drive connection between member 128 and the transmission casing, thereby holding against rotation and releasing sun gear 62.

Figure 2 contains a schedule showing the engaged and disengaged states of friction clutches 118, 120, 122 and brakes 124, 126, and the driving and overrunning condition of one-way coupling 82.

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The first speed ratio in the drive condition results by engaging clutch 118. This causes coupling 82 to drive and thereby to hold carrier 66 and ring gear 56 against rotation on the transmission housing. The sun gear 54 of the first gear unit is driven through the chain drive mechanism 30, ring gear 56 is held fixed against rotation through clutch 118 and coupling 82, and the output is taken at carrier 56 and shaft 90. The final drive gearset 20 25 30 35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140 145 150 155 160 165 170 175 180 185 190 195 200 205 210 215 220 225 230 235 240 245 250 255 260 265 270 275 280 285 290 295 300 305 310 315 320 325 330 335 340 345 350 355 360 365 370 375 380 385 390 395 400 405 410 415 420 425 430 435 440 445 450 455 460 465 470 475 480 485 490 495 500 505 510 515 520 525 530 535 540 545 550 555 560 565 570 575 580 585 590 595 600 605 610 615 620 625 630 635 640 645 650 655 660 665 670 675 680 685 690 695 700 705 710 715 720 725 730 735 740 745 750 755 760 765 770 775 780 785 790 795 800 805 810 815 820 825 830 835 840 845 850 855 860 865 870 875 880 885 890 895 900 905 910 915 920 925 930 935 940 945 950 955 960 965 970 975 980 985 990 995 1000 1005 1010 1015 1020 1025 1030 1035 1040 1045 1050 1055 1060 1065 1070 1075 1080 1085 1090 1095 1100 1105 1110 1115 1120 1125 1130 1135 1140 1145 1150 1155 1160 1165 1170 1175 1180 1185 1190 1195 1200 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56 are drivably connected through member 86 and clutch 118, and the output of gear unit 50 is taken at ring gear 64, carrier 58, and output shaft 90. The third forward speed ratio is an underdrive ratio. The torque reaction is provided at ring gear 70, which is fixed on the transmission housing against rotation.

The fourth forward speed ratio is approximately a direct drive ratio that results by maintaining clutch 118 engaged, releasing clutch 122, and engaging clutch 120. This action causes coupling 82 to overrun.

The third planetary gear unit 52 overdrives ring gear 72 in relation to the speed of input shaft 44. Torque output from gear unit 52 is transmitted through clutch 120, carrier 66, member 86, and clutch 118 to ring gear 56 of the first planetary gear unit 48. Sun gear 54 is driven at the speed of the input shaft 44, and the output is taken at ring gear 64, carrier 58, and output shaft 90. Torque reaction is provided at ring gear 70, which is held fixed against rotation on the transmission housing.

An upshift to the fifth forward speed ratio results when clutch 118 is disengaged, clutch 120 is maintained engaged, and clutch 122 is engaged. When these friction elements are fully engaged and disengaged, coupling 82 overruns and output shaft 90 is overdriven in relation to the speed of shaft 44. In this instance, two components of planetary gear units 48 and 50 are driven at different speeds through the third planetary gear unit 52. Input shaft 44 is drivably connected through carrier 74, clutch 122, and member 128 to ring gear 62 at the same speed as that of input shaft 44, but a speed reduction occurs in gear unit 52 such that ring gear 72 overdrives carrier 66 through clutch 120. Sun gear 54 is drivably connected directly to input shaft 44 so that both sun gears 54, 62 are driven at the same speed and carrier 66 is overdriven. The output of the gear units is taken at sun gear 64, carrier 58, and output shaft 90.

An upshift to the sixth forward speed ratio results by maintaining clutch 120 engaged, disengaging clutch 122, and engaging brake 126. With the friction elements so disposed, coupling 82 overruns. In this speed ratio, ring gear 54 is driven directly from input shaft 44, but carrier 66 is overdriven through clutch 120 by third gear unit 52, whose carrier 74 is driven from input shaft 44 and whose ring gear 70 is fixed against rotation. Sun gear 62 of the second gear unit 50 is fixed against rotation on the transmission housing through engagement of brake 126; therefore, a further speed reduction occurs in gear unit 50, and its ring gear 64 drives the output carrier 58 and output shaft 90.

Referring now to Figures 3 and 4, brake 220, which is an hydraulically-actuated friction brake, holds sun gear 70 against rotation on the housing 80 when the brake is applied and allows that gear to rotate when the brake is released. Therefore, sun gear 70 is not fixed permanently against rotation, as Figure 2 shows. Furthermore, clutch 120 of Figure 1 is deleted and replaced by a permanent drive connection between ring gear 22 and carrier 66.

The embodiment of Figure 3 operates essentially the same in the three lowest forward gears as the embodiment of Figure 1 described above. An upshift to fourth gear results by maintaining clutch 118 engaged and engaging brake 220. This action causes coupling 82 to overrun, holds sun gear 70, and produces a torque reaction on the housing at brake 220. Gear unit 52 overdrives carrier 66 in relation to the speed of input shaft 44. Sun gear 54 is driven at the speed of input shaft 44, and ring gear 56 is driven at the speed of carrier 66, due to the engagement of clutch 118.

The upshift to fifth gear from fourth gear results by maintaining brake 220 applied, disengaging clutch 118, and engaging clutch 122. Sun gear 62 is driven directly by input shaft 44 through carrier 74, clutch 122, and member 128, and carrier 66 is overdriven in relation to the speed of shaft 44 through gear unit 52. The output is taken at ring gear 64, carrier 58, and shaft 90.

Sixth gear is produced when brake 124 and clutch 122 are disengaged. Carrier 66 is overdriven at the speed of ring gear 72 due to the torque reduction accomplished in gear unit 52. Sun gear 54 is driven directly from shaft 44, and the torque reaction is provided at sun gear 70, which is held on the housing at brake 220. Ring gear 72 overdrives carrier 66 and sun gear 62 is held against rotation by brake 126. The output is taken at ring gear 64, carrier 58, and shaft 90.

Reverse drive results when clutch 122 and brake 124 are applied. Sun gear 62 is driven at the speed of input shaft 44 through clutch 122 and gear unit 52, carrier 66 is held due to the engagement of brake 124, and the underdriven output is taken at ring gear 64, carrier 58, and shaft 90.

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Claims

1. A multiple speed ratio automatic transmission, comprising:

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an input shaft (44);
 an output shaft (90);
 first (48), second (50), and third (52) gear units, each gear unit having a sun gear, ring gear, planet pinions meshing with the sun gear and ring gear, and carrier rotatably supporting the planet pinions;
 the sun gear (54) of the first gear unit (48), carrier (74) of the third gear unit (52) and input shaft (44) drivably connected mutually; the carrier (58) of the first gear unit (48), ring gear (64) of the second gear unit (50) and output shaft (90) drivably connected mutually; the carrier (66) of the second gear unit (50) and ring gear (72) of the third gear unit (52) drivably connected mutually;
 an overrunning brake (82) for holding the carrier (66) of the second gear unit (50) against

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rotation in one rotary direction and permitting rotation in the opposite direction;
 a first clutch (118) for releasably connecting the ring gear (56) of the first gear unit (48) and carrier (66) of the second gear unit (50);
 a second clutch (122) for releasably connecting the carrier (74) of the third gear unit (52) and sun gear (62) of the second gear unit (50);
 a first brake (124) for releasably holding the carrier (66) of the second gear unit (50) against rotation;
 a second brake (126) for releasably holding the sun gear (62) of the second gear unit (50) against rotation; and
 means for holding the sun gear (70) of the third gear unit (52) against rotation.

2. A transmission as claimed in claim 1, wherein the means for holding the sun gear (70) of the third gear unit (52) against rotation serve permanently to fix the ring gear (70) of the third gear unit (52) against rotation on the housing (80) of the transmission and the ring gear (72) of the third gear unit (52) is releasably connected by means of a third clutch (120) to the carrier (66) of the second gear unit (50).

2. A transmission as claimed in claim 1, wherein the means for holding the sun gear (70) of the third gear unit (52) against rotation comprise a third brake (220) for releasably holding the ring gear (70) of the third gear unit (52) against rotation and the ring gear (72) of the third gear unit (52) is permanently connected for rotation with the carrier (66) of the second gear unit (50).

4. A transmission as claimed in any preceding claim, further comprising:

a torque converter (10) mounted for rotation about an axis parallel to the input shaft (44), adapted for connection to a power source, for producing an hydrokinetic drive connection between the input shaft (44) and power source; and
 a chain drive mechanism (30) for transmitting power between the power source and input shaft (44).

5. A transmission as claimed in any preceding claim, wherein:

the third gear unit (52) directly connects the input shaft (44) and sun gear (62) of the second gear unit (50) when the second clutch (122) is released; and
 the third gear unit (52) underdrives the carrier (66) of the second gear unit (50) in relation to the speed of the input shaft (44) when the sec-

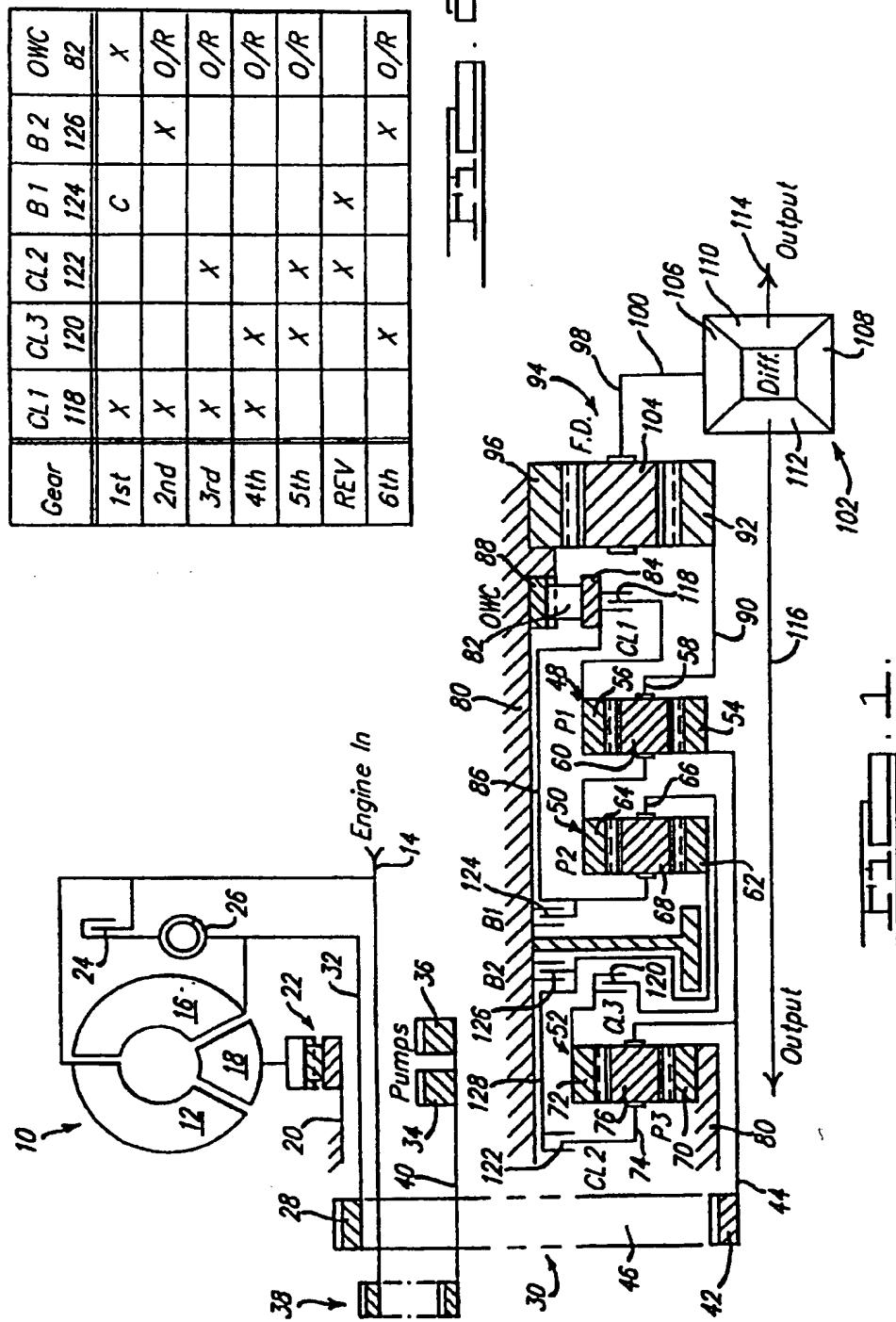
ond clutch (122) is engaged.

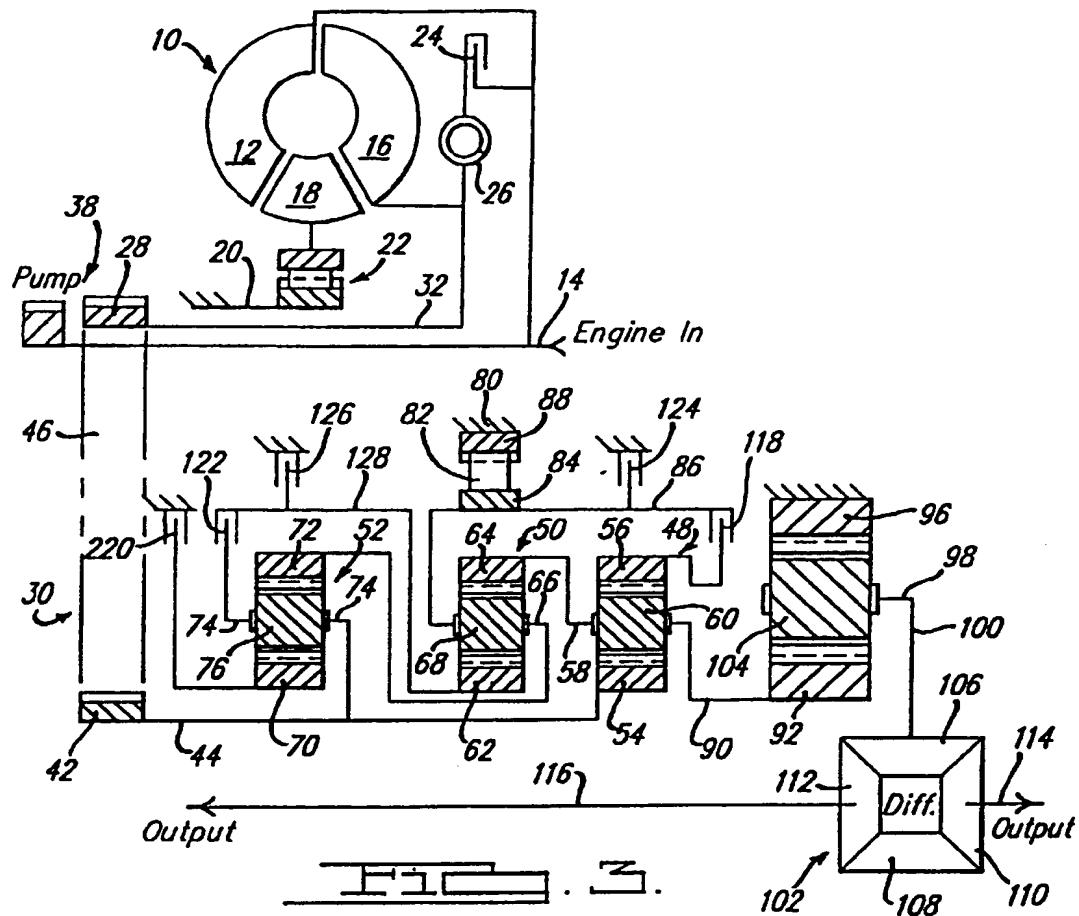
6. A transmission as claimed in any preceding claim, wherein the overrunning brake means (82) comprises:

an overrunning coupling having a first race (88) fixed against rotation;
 a second race (84) concentric with and radially spaced from the first race (88); and
 a coupler located between the races (84, 88) for producing a one-way drive connection therebetween and permitting free rotation in the opposite rotary direction.

7. A transmission as claimed in claim 5, wherein:

the carrier (66) of the second gear unit (50) is drivably connected to the second race (84) of the overrunning brake (82);
 the first clutch (118) releasably connects the ring gear (56) of the first gear unit (48) to the second race (84) of the overrunning brake (82) and carrier (66) of the second gear unit (50); and
 the first brake (124) releasably holds the carrier (66) of the second gear unit (50) and second race (84) of the overrunning brake (82) against rotation.





| Range | 220 | 118 | 122 | 124 | 126 | 82 |
|-------|-----|-----|-----|-----|-----|-----|
| 1st | | X | | X | | X |
| 2nd | | X | | | X | O/R |
| 3rd | | X | X | | | O/R |
| 4th | X | X | | | | O/R |
| 5th | X | | X | | | O/R |
| 6th | X | | | | X | O/R |
| REV | | | X | X | | |

— E — . 4.

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